

THE DESIGN AND CALIBRATION OF A
VISCOUS AIR FLOW METER

By

S. VICAR HUSAIN

Bachelor of Science

Nizam College

Hyderabad, India

1947

Bachelor of Engineering

Osmania University

Hyderabad, India

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VISCOUS AIR FLOW METER

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THESIS APPROVED:

W. H. Easton

Thesis Adviser

J. A. Boggs

Faculty Representative

Robert Markier

Dean of the Graduate School

349746

PREFACE

Engineers who deal with research and development work on machines through which there is air flow are ever conscious of the need for improved devices for measuring air flow - particularly in cases where the flow is not steady. Thus the inadequacy of existing equipment present a challenge to the individual to improve existing designs or develop new and improved designs.

The need for an improved device for measuring the air flow to an internal combustion engine prompted the author to design and calibrate the viscous air-flow meter dealt with in this report.

Throughout the report it is assumed that the reader is familiar with the fundamentals of hydraulics as applied to flow problems. The entire report can be of value to those who are concerned with air flow measurement under pulsating or steady flow conditions. The author now wishes to express his sincere thanks to Professor W. H. Easton for carefully reading the entire report and offering valuable suggestions and comments, and to Professor B. S. Davenport for his valuable advice and practical suggestions. I also wish to express my special appreciation and thanks to Professor Gordon Smith for his valuable help in machining the nozzles and other pieces of apparatus used during the test.

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SYMBOLS AND ABBREVIATIONS

A	area - flow or cross sectional, square inches
C	coefficient - constant for the nozzle
D	diameter in inches
E	energy in general
F	force
F	degrees Fahrenheit
g	acceleration due to gravity feet per sec per sec
h	head
h_L	head loss
h_s	static head
h_t	total head
h_v	velocity head
J	Joules equivalent
k	ratio of specific heats C_p/C_v
c	specific heat
K	flow coefficient (coefficient of discharge with approach factor included)
L	length of the element
M	molecular weight
m	hydraulic mean radius
p	pressure, psia
Q	volumetric rate of flow
r	radius in general
R	gas constant
t	temperature Fahrenheit

V velocity feet per seconds
 v volume in general
 W flow rate lbs. per seconds
 X unknown quantity.
 Y expansion factor for the nozzle.
 Z Elevation of the center of gravity of any section
 Z Temperature correction factor

Abbreviations

BTU British thermal units
 ASME American Society of Mechanical Engineers
 BWG Birmingham wire gage
 bhp brake horse power
 cc cubic centimeters
 ft feet.
 sq square.
 sec seconds
 hr hour
 lb pounds
 psi pressure pounds per square inch
 fps feet per second.
 in. inches
 min minute
 psia pounds per square inch absolute.
 rpm revolution per minute.
 Ra or Rd Reynold's number dimensionless
 dx/dy differential of x by differential of y

Greek Letters

Δ change of value in general

μ absolute viscosity

ρ density

Σ summation of values

CHAPTER I

INTRODUCTION

The measurement of the quantity of air is a vital factor in research work dealing with carburetors and internal combustion engines.

The maximum power an internal combustion engine can develop is dependent on the amount of air it can take in. The power and efficiency are also affected by the air-fuel ratio.

Until recently, there were two principal methods for measuring air consumption. The first method is the air bell method; a device employed for many tests. Although it gives an accurate determination, it has several drawbacks. This type of unit is very large; requires extensive piping and is not portable. The second method uses elements of "kinetic type," such as orifices, venturi, and nozzles. They are simple and handy to use, but are rather inaccurate due to the "root mean square" errors unless the flow is quite steady. The errors can be reduced in all the above cases by means of smoothing capacity tanks, but the size of tanks required may be too large to be practical.

The recognition of these drawbacks lead to the design of the "viscous flow air meter." This type of apparatus was first designed by J. F. Alcock. The principle on which this instrument is based is, that the flow across a viscous element is proportional to the pressure drop. Thus this type of meter does not have the errors due to "root mean square." As there is no literature available regarding the design and experimental results for this type of meter, the author of this paper undertook the problem of designing and calibrating a viscous flow air meter. The

calibration of the viscous flow air meter is done by use of standard nozzles described in the ASME power test codes.

CHAPTER II

REVIEW OF LITERATURE

There are two broad fields in the study of fluids: hydrodynamics, dealing with the motion of fluids, and hydrostatics, dealing with fluids at rest. Fluids are generally divided into gases and liquids; although the theory of fluids can be applied to both. The chief distinctions to be made are in density and compressibility. The smaller density and greater compressibility of a gas allows it to fill the volume of any vessel in which it is confined, whereas a small quantity of liquid in a large vessel presents a free surface. Both types of fluid show resistance to motion in which a further property called viscosity is involved. In the study of the motion of fluids, the physical properties of matter in the fluid state which have to be considered are: density, compressibility, surface tension, viscosity and shear elasticity.

The actual motion of fluids results from the difference of pressure, or of density. The fluid, whether liquid or a gas, flows in the direction of a gradient of pressure, and from a place where the fluid is compressed to one where it is rarefied until equilibrium is attained. Continuous motion will occur as long as the unbalanced pressure difference is maintained by any of the external forces or difference in density.

This paper deals with hydrodynamics, or the motion of fluid; specifically the flow of air has been chosen for consideration.

Before going into the details of the flow problems of air, it is necessary to know the types of flow encountered in engineering investigations. The most common method of determining the nature or the type

of flow of fluids is by performing simple experiments which show that there are two entirely different types of flow in a pipe. These simple experiments show the nature of flow in pipes, one being viscous and the other turbulent, also with conditions existing between the two. By the experiments performed with the flow of water in a glass tube, it has been shown that if color is introduced into the stream and the velocity of the main stream is varied, the stream of colored water flows in a straight path at low velocities, and if the velocity is increased, the colored filaments show a tendency to break up at a certain "critical velocity." At higher velocities the filaments no longer flow in straight paths, but are dispersed throughout the entire amount of fluid. The type of flow that occurs at velocities below the "critical" is known as viscous, laminar, or streamline flow. Such a type of flow is characterized by the shearing of concentric cylinders of the fluid past one another in an orderly fashion. The velocity of the fluid is greatest at the center of the pipe and decreases sharply to zero at the outer wall surface. At velocities greater than the critical velocity the flow is turbulent. In turbulent flow, there is random motion of the fluid particles in the direction transverse to the direction of the main flow. Even though there is turbulence throughout a great portion of the fluid in the pipe, there is always a thin layer known as the "boundary" layer or laminar sub-layer of fluid at the pipe wall which has viscous flow. A number of previous experiments, as well as theoretical considerations, show that the nature of flow in pipes, whether it is viscous or turbulent, depends upon the pipe diameter, density, velocity of flow, and the viscosity of the flowing fluid. The numerical value of the

dimensionless combination of these four variables is known as Reynold's number. This number indicates whether the flow is viscous or turbulent. Reynold's number can be stated in the form of an equation as:

$$R_e = \frac{DV\rho}{\mu}$$

By experiment it has been found that the flow is usually viscous for Reynold's numbers less than 1200 and turbulent for Reynold's numbers greater than 2200. Between these two values lies a transition region in which the flow may be either viscous or turbulent, depending upon the condition of flow as it enters the section being considered and upon the roughness of the pipe walls.¹

Another determining factor is friction, accompanying any fluid flow in the pipes. There are two types of friction: the internal friction caused by the rubbing of the fluid particles against each other, and the other is external friction caused by the rubbing of the fluid particles against the pipe wall. In both cases, energy loss due to friction is converted into heat energy. This heat energy may be entirely absorbed by the fluid in adiabatic flow or it may be entirely dissipated through the pipe walls in isothermal flow. In compressible fluids some of the frictional heat may be absorbed in accelerating the fluid and yet the flow may be isothermal. In the case of the flow of liquids and gases at atmospheric temperatures through pipes, we can assume the flow to be isothermal.

Adiabatic flow is generally assumed to take place in nozzles, orifices, and short tubes through which the fluid is moving at high

¹ "Flow of Fluids Through Valves, Fittings, and Pipe", Engr. and Research Division, Crane Company, Chicago, Ill., Tech. paper No. 409, May, 1942.

velocities. A number of different methods of flow measurement are available; the experimenter must choose from the available information the one that is best suited to the particular purpose. The selection of a suitable method for measuring flow should be governed by the simplicity, directness, and the degree of accuracy required. A direct method of obtaining the flow is to collect and weigh the quantity of fluid delivered in a given time. This method is simple and is quite easily applied to the flow measurement of a fluid whose density is higher than that of gases. However, for the flow measurement of gases the most desirable method generally involves the measurement of some physical effect associated with the flow. Three physical effects that have been found by experience to be suitable for the flow measurement of gases are: the pressure associated with the motion; mechanical effects, such as the rate of rotation induced in light vanes suitably mounted in the stream; and the rate of cooling of a hot body, such as an electrically heated wire introduced into the air current. Of the three above mentioned methods, the first is of the greatest importance. A properly designed instrument suitably inserted in the stream measures a pressure commonly called the "velocity head", which is a characteristic of the motion. It can be measured with a suitable pressure gauge. These instruments can be used for the measurement of air speed. The anemometer, which is a mechanical device for measuring air speed, depends for its action either on mechanical or electrical effects and requires calibration against a standard instrument. These pressure measuring instruments are known as pressure tube anemometers.

Other devices which involve the measurement of pressure difference are orifice plates, venturi tubes, and nozzles. All these measure the

pressures, not the velocity head, which depend on the dimensions and form of the instruments themselves, as well as on the motion of the fluid that passes through them. These have to be calibrated before they can be used as standards. However, the ASME has standardized nozzles of various sizes which are used in all power test codes in this country.

Detailed descriptions of mechanical anemometers and hot wire anemometers will not be given in this paper. However, some of the instruments of the above types are convenient for practical use and have a comparatively wide field of application.

As mentioned previously, the orifice and the venturi are the most used devices for measuring the flow of fluids. The chief disadvantages of the orifice plate are the element of uncertainty regarding the area of the vena contracta and the large resistance it offers to the passage of air through it. Both of these undesirable features have been avoided by the use of venturi, but only at the expense of considerably increased cost and installation difficulties. A well shaped nozzle, which is, in effect, a venturi with the expansion cone removed, is a device intermediate between the orifice and the venturi. The unrecoverable loss of pressure through a nozzle is slightly less than that of the orifice plate, but, in this respect, it is inferior to the venturi. In regard to cost and ease of installation, it is superior to the venturi.

The "Vereiu Deutscher Ingenieure" has standardized this form of nozzle under the title of "German Standard Nozzle 1930." The ASME has also standardized various sizes of nozzles; and recommends their use

for flow tests of various machines.

Any form of restriction will generate a pressure differential which can be used as a measure of flow rate. If a non-standard primary element is to be used for flow measurement, calibration is required to establish the value of the discharge coefficient. This discharge coefficient will not be constant in such a case over the entire flow range due to the viscous forces, or non-uniform velocity distribution. Charts and tables are available in the ASME test code for use in evaluating the discharge coefficients for ASME standard nozzles for any existing flow conditions. Hence, the standard ASME nozzles were selected for use in calibrating the viscous air flow meter. This instrument can be used with pulsating flow and does not require the assistance of a large smoothing capacity in series. It can therefore be used on either single or multi-cylinder engines. Alcock's viscous air flow meter proved to be very useful in Alcock's own laboratory. It was designed for measuring the scavenge air consumption of high speed, two stroke research units, and the air consumed by highly supercharged units, as well as for engines of moderate output.

There is very little information on the design, tests, and calibration of Alcock's meter available in the library; except for a brief review of the apparatus found in Ricardo's Internal Combustion Engines. The Alcock viscous air flow meter consists essentially of a honey-comb arrangement of narrow equilateral triangular passages 3 inches long by .017 inches in height. The air flow through the passages is viscous for flow rates within the working range of the meter.¹ The

¹ H. R. Ricardo, High Speed Internal Combustion Engines, Vol. II., London, 1923.

resistance offered is therefore proportional to the velocity. The "root mean square" error of the kinetic type of meter; such as the sharp edge orifice, venturi, or nozzle is eliminated by Alcock's meter. The resistance is measured by a manometer connected across the "meter element." By special design of the manometer connections, errors due to the flow in and out of the connections set up by pressure variation in the line are avoided. The air passes through a cleaner and then through the meter element. The meter element is constructed of alternate layers of flat and corrugated sheet metal strips wound around a core. The manometer connections are tubes spanning the face of the element and having a single row of holes in each. In the upstream tube the holes point in a downstream direction. In the downstream tube the holes point upstream. The reason for such an arrangement is to provide a reverse kinetic head, which automatically corrects the small kinetic pressure loss in the element due to entry effects.

The pressure drop across the elements was found to be proportional to the gas velocity. In the case of pulsating flow in single cylinder engines, the accuracy of measurement was unimpaired. This meter was calibrated under steady flow conditions by comparing it with a "German standard nozzle" which had been checked against a "gas holder." The desirable features of the instrument are that the pressure differential is proportional to rate of flow and a satisfactory accuracy is obtained over a wide range of flow.

CHAPTER III

STATEMENT OF PROBLEM

The main purpose of this investigation was to design and calibrate an air-flow meter suitable for measuring air flow under pulsating flow conditions. It is intended primarily for the measurement of the flow rate of air supplied to an internal combustion engine. To attain this objective, an air flow meter was designed on the basis of the theory that whenever there is viscous flow of a fluid across any primary element, the rate of flow is directly proportional to the pressure drop across the element. This is true at low values of Reynold's numbers. Based on this theory and the previous experience of Alcock, the author designed a viscous flow meter. The details of construction are given in the following pages. Standard ASME nozzles were used in the calibration of meter.

CHAPTER IV

ANALYSIS OF THE PROBLEM AND THEORY

The theory on which the design of the viscous flow meter is based involves the relationship between the flow rate and differential pressure under special flow conditions. These conditions are governed by the viscosity, which in turn depends upon the temperature of the fluid. The Reynold's number may be used to determine the range of flow rate over which laminar flow may be expected.

As the viscous flow meter was calibrated by use of standard nozzles, an explanation of the equations used is presented in the appendix. The "National Instrument Laboratories" uses the following equation to express the relationship between the flow rate and differential pressure across the element of a viscous flow meter.

$$\Delta p = a \rho Q^2 + b \mu Q$$

where Q is the volume flow rate

μ is the viscosity

ρ is the density

a and b are constants

It follows from the above equation that if a flow element is designed and constructed so that $a = 0$, then the differential pressure for a particular fluid will be directly proportional to the volume flow rate.

This condition of flow will exist only for low values of the Reynold's number.

The National Instrument Laboratories, Incorporation uses the Reynold's number equation as follows.

$$R_e = \frac{2 \rho Q}{\mu S}$$

where S equals the thickness of the flow channel.

If the flow conditions are such that $b = 0$, then the pressure differential is proportional to the square of the volume flow rate. This is the condition of flow through orifices and nozzles operating at a high Reynold's number, and the flow is considered to be turbulent. With an orifice operating at a high Reynold's number, flow rate errors due to viscosity become large. Similarly, if a laminar flow meter is used for flows above its design range, the errors due to the density of the gas becomes an important factor. In designing the viscous flow meter the flow channels of the viscous element are kept identical in order to obtain the desired degree of accuracy. It has been found by the National Instrument Laboratories, Incorporation that a high degree of accuracy is obtained in their flow meter when the Reynold's number at full scale flow is less than 400.¹

Theory of Viscous Flow

The absolute viscosity of a fluid is defined as its internal resistance to flow. In viscous or laminar flow, the internal resistance is due primarily to the cohesive shearing stress existing between the layers or laminae of infinitesimal thickness constituting the fluid stream. The shearing stress is greater in fluids of high viscosity than in those of low viscosity.

For all practical purposes, it can be assumed that viscosity is independent of variations in pressure but varies with temperature. In liquids an

¹ N. I. Laboratories, Inc., "Vol-O-Flo meter for Gas Measurement"

increase in temperature causes a decrease in viscosity, in gases an increase in temperature causes an increase in viscosity.

The modern theory of viscosity was first advanced by Sir Isaac Newton in his monumental work "Principia" and is generally known as Newton's hypothesis of viscous flow.

In mathematical terms, Newton's Theory states that the viscosity of a fluid is equal to the shearing stress divided by the rate of shear.

$$\mu = \frac{S}{\Delta v / \Delta y}$$

where

μ is the absolute viscosity of fluid.

S is the unit shearing stress, F/A or Force/Area.

$\Delta v / \Delta y$ is the rate of shear or change in velocity with respect to the distance between shearing surfaces.

The above formula can be illustrated as follows:

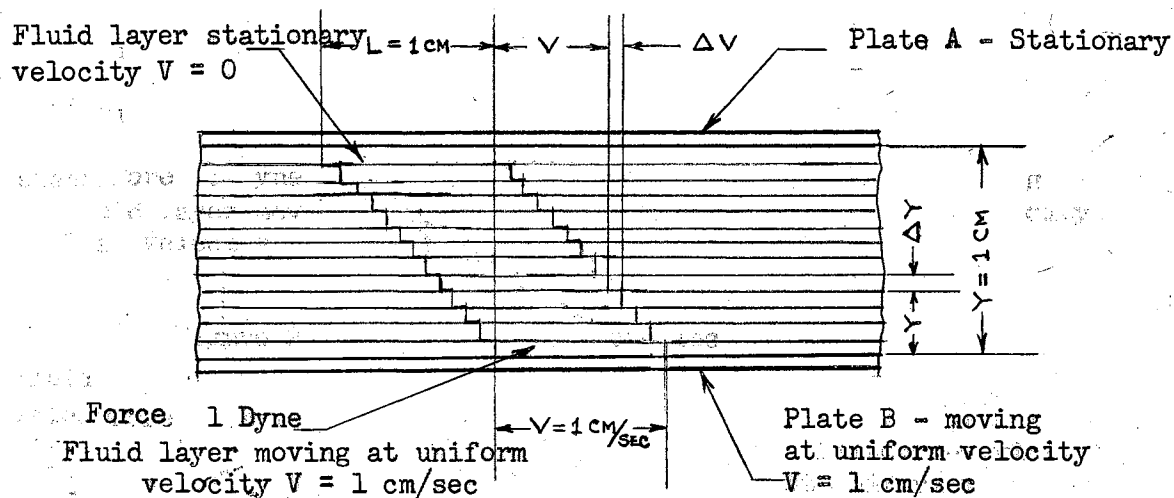


Figure 2. Newton's Theory of Viscous Flow

Consider an elementary unit section of a viscous fluid filling the space between two parallel plates A and B in Figure 2. Assume the transverse distance between the two plates as unity. Plate A is stationary and plate B is moving with a constant velocity, V . The distance between the two plates is represented by Y .

Assume further that the fluid is in a state of steady flow with each layer ΔY , moving at its individual constant velocity $V + \Delta V$, and with no layer subject to acceleration or deceleration. The fluid layer or film closest to the surface of either plate remains fixed to the plate by adhesion and therefore has no motion relative to the plate. That layer of infinitesimal thickness closest to surface A remains fixed to plate A and does not move. Likewise, the fluid layer closest to the surface B is held fast to plate B by adhesion and has no motion relative to the plate B. However, if plate B is moving with a constant velocity V , relative to plate A, then the fluid layer adhering to plate B also moves with constant velocity V relatively to the fluid layer adhering to plate A. It is therefore evident that no shearing action takes place between the fluid and the plate to which it adheres, but takes place only between the fluid layers.

It is assumed that the fluid layers are of the same size and shape, and the shearing stress between any two layers or the adhesion between the fluid and the plates is equal and opposite to the force, F , producing flow.

$$\text{i.e. } S = F$$

S is the total shearing stress

F is the force producing flow

If s is the unit shearing stress between any two fluid layers, then

s equals F/A .

A is equal to the shearing area of fluid layer.

Then,
$$\mu = \frac{F/A}{\Delta V/\Delta Y}$$

From Figure 2, it is evident that the sum of ΔY is equal to Y and the sum of the elementary velocity. ΔV is equal to V .

therefore,
$$\mu = \frac{F/A}{V/Y}$$

 or
$$= \frac{F/L^2}{(L/T)/L} = \frac{FT}{L^2} = \frac{\text{Force} \times \text{Time}}{\text{Length}^2}$$

If CGS units are used in the above formula, the unit of absolute viscosity will be the poise. Thus the poise is the force in dynes per sq cm at a velocity of one cm per sec at a distance of one cm.

Viscous Flow in Capillary Tubes

"The work of the French physician and scientist, Poisseuille, in whose honor the unit of absolute viscosity, the "poise", was named, did much to confirm and illuminate Newton's hypothesis of viscous flow in capillaries or small tubes. Poisseuille's experiments made in about the year 1840 were undertaken in an effort to determine the nature of the flow of blood in the human vascular system. This experimental work was not done intentionally for the purpose of confirming a theory of fluid mechanics, but that is exactly what it did. Unfortunately, it was not until after his death, when a mathematical analysis was made of his findings, that the real value of Poisseuille's work as a major contribution to fluid mechanics was discovered. Poisseuille's Law in the light of mathematical analysis forms the basis of the modern theory of viscous flow in tubes, capillaries and pipes."

2 McClain, Clifford. "Fluid Flow in Pipes", New York Industrial Press, 1952.

It is interesting to study in detail Poisseuille's Law. A small capillary tube of radius (r) and length (L) is shown in the figure on the next page, through which a viscous fluid is flowing. It is assumed that the fluid is in a state of steady flow, that is to say the layer is moving at constant velocity, and no layer is subjected to acceleration or deceleration. Under such conditions the fluid is in equilibrium with respect to external forces. It is assumed that the layer of the fluid adhering to the surface of the tube has zero velocity and the maximum velocity is at the center of the tube. The pressure (P) existing at the entrance or upstream end of the tube is entirely dissipated in overcoming the flow resistance throughout the entire length (L) resulting in zero pressure at the exit or downstream end of the tube. Since the pressure drops at a constant rate per unit of length throughout the tube, the unit pressure drop is equal to P/L .

The flow in a tube can be assumed to consist of an infinite number of hollow concentric telescopic cylinders of unit length and with wall thickness dy .

The constant velocity of individual concentric cylinders is progressively greater toward the center of the tube by a small increment dv . The flow rate past any transverse section of the stream must be constant. Otherwise, the flow would be distorted and result in turbulence.

If one cylinder is farther downstream than another, the pressure exerted on its upstream end is less than that exerted on the other cylinder in proportion to the distance by which it leads the other cylinder at a given time. There would be a similar difference in the pressure exerted on the downstream end of the cylinder. The ends of the fluid

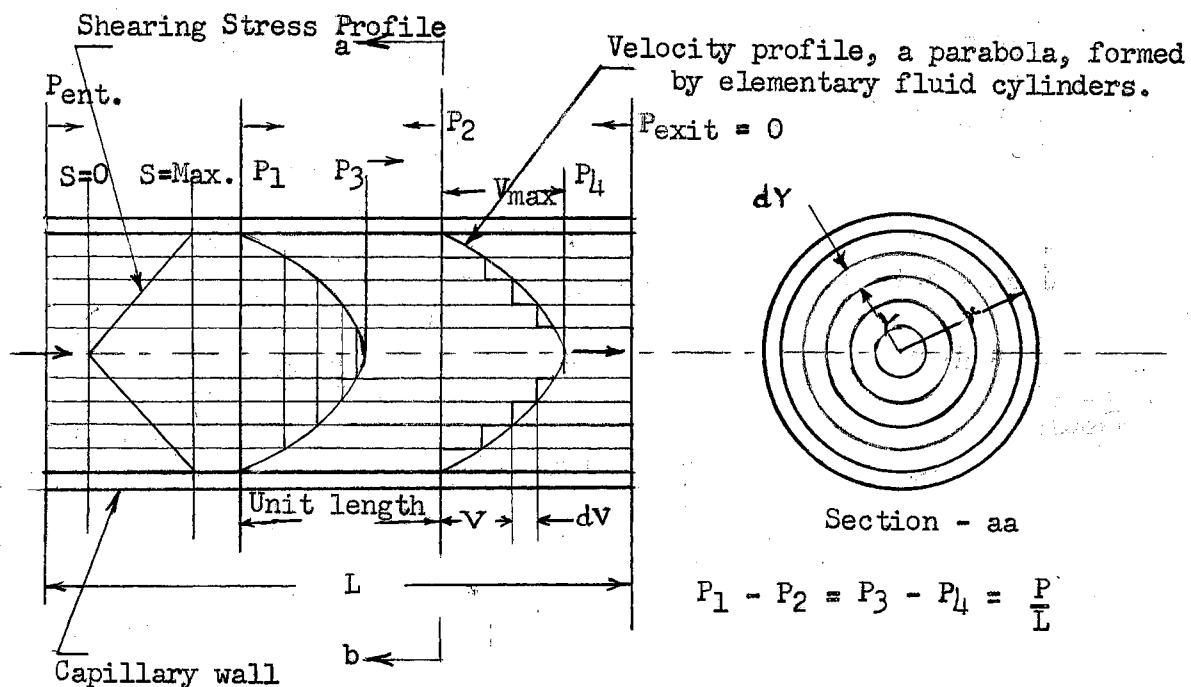


Figure 3. Viscous Flow in Capillary Tube

cylinders have the shape of an annular ring with a wall thickness of (dy) . Therefore, by equating the equal and opposite forces and using the theorem of Pappus for finding the area of the ends of the hollow cylinders for any cylinder of unit length

$$2 \pi y \frac{P}{L} dy = 2 \pi s (y + dy)$$

for the entire length L of the tube

$$2 \pi y P dy = 2 \pi s (y + dy) L$$

for the entire stream

$$\int 2 \pi y P dy = \int 2 \pi s (y + dy) L$$

$$\pi y^2 P = 2 \pi y s L$$

therefore shearing stress $s = \frac{\pi y^2 P}{2 \pi y L} = \frac{y P}{2L}$

From the equation obtained for the shearing stress it can be seen that the shearing stress varies with the distance (y) from the axis of the tube.

In the case of simple viscous flow, the shearing stress is the same for each fluid layer and is independent of the distance (y).

Newton's expression for unit shearing stress in a differential form is

$$s = -\mu \frac{dV}{dy}$$

The reason for the negative sign is that y is measured from the center of the tube and not from the outer edge of the tube.

Therefore

$$-\mu \frac{dV}{dy} = \frac{yP}{2L}$$

$$\text{rate of shear} = \frac{dV}{dy} = -\frac{yP}{2L\mu}$$

From this expression it can be shown that the rate of shear is not constant for each fluid cylinder, but varies with y when P , L , and μ remain constant. In simple viscous flow the rate of shear is the same for each fluid layer. The velocity changes as a uniform or straight line function and the shearing areas are the same for each layer.

It is desirable now to find an expression for the velocity of any fluid cylinder at any distance y from the axis of the tube, and then a general equation for viscous flow in a tube can be established

$$dV = -\frac{yP}{2L\mu} dy$$

$$\text{therefore} \quad V = -\frac{y^2P}{4L\mu} + C$$

where C is a constant of integration.

Since the velocity is zero at the tube walls when y equals the radius of the tube.

$$C = \frac{r^2 P}{4L\mu}$$

$$V = \frac{P}{4L\mu} (r^2 - y^2)$$

Hence, it is established that the velocity gradient of the telescopic fluid cylinders takes a form of a solid parabola within the tube shown in Figure 3. The volume of fluid flowing through the tube during an interval of time can now be obtained. The above expression gives the velocity of the fluid cylinder located at a distance (y) from the axis of the tube in terms of distance per second. Since the distance travelled in unit time is the same as the velocity, the total volume flowing in t sec, will be the sum of all these elemental volumes multiplied by t.

Thus volume = area x velocity x time

$$dv = 2\pi y dy \times V \times t$$

$$= 2\pi y dy \frac{P}{4L\mu} (r^2 - y^2) \times t$$

$$v = \int_0^r \frac{\pi Pt}{2L\mu} (r^2 - y^2) y dy$$

$$v = \frac{\pi Pt}{2L\mu} \left[\int_0^r r^2 y dy - \int_0^r y^3 dy \right]$$

$$= \frac{\pi Pt}{2L\mu} \left[\frac{r^2 y^2}{2} - \frac{y^4}{4} \right]_0^r$$

$$= \frac{\pi Pt}{2L\mu} \left[\frac{r^4}{2} - \frac{r^4}{4} \right]$$

$$= \frac{\pi Pt r^4}{8L\mu}$$

$$\therefore \mu = \frac{\pi Pt r^4}{8L v}$$

The driving force in terms of fluid head is $h = P/w$

where h is the head in feet of flowing fluid

w is the specific weight of the fluid equal ρg

ρ is the density of the flowing fluid

g is the acceleration of the gravity

Therefore,

$$\mu = \frac{\pi \rho g h r^4 t}{8 L v}$$

The volumetric flow rate of an incompressible fluid can be expressed as

$$Q = \frac{v}{t} \quad \text{or} \quad v = Q t$$

Q is the volumetric flow rate cubic units/sec

v is the volume of fluid flowing in t sec

t is the duration of flow in sec

therefore,

$$\mu = \frac{\pi \rho g h r^4}{8 L Q}$$

or

$$Q = \frac{\pi \rho g h r^4}{8 L \mu}$$

According to the law of continuity, the quantity of fluid flowing per second through any pipe is expressed as $Q = VA$.

V is the mean velocity of fluid or units of length per sec. For a round pipe, the above formula becomes

$$Q = V \pi r^2$$

Therefore,

$$V \pi r^2 = \frac{\pi \rho g h r^4}{8 L \mu}$$

or

$$h = \frac{8 L V \mu}{\rho g r^2}$$

where $h = (h_1 - h_2) = h_L$

h_L = head lost in producing flow against viscous resistance.

In deriving the fundamental equation stated above, it is assumed that the entire pressure head is converted into velocity head. Thus

$$h_2 = 0 \quad h = h_1 = h_L$$

Usually the above equation is written in terms of the pressure differential for the section of pipe being considered.

It can be written: $\Delta p = \rho g h$

where Δp is the pressure drop across a transverse section of the pipe under consideration causing the flow of the fluid. Thus

$$Q = \frac{\pi \Delta p r^4}{8 L \mu}$$

This equation for the flow is developed for circular cross section pipe. The same equation can be applied to any other section, if only a factor known as hydraulic mean radius is taken into account. Therefore, the above equation can be expressed in terms of hydraulic mean radius (m) to be applied to different shape sections,

$$m = \frac{\text{cross sectional area}}{\text{wetted perimeter}}$$

For a circular pipe

$$m = \frac{\pi r^2}{2\pi r} = \frac{r}{2}$$

where r is the radius of the pipe, or $r = 2m$.

Therefore,

$$Q = \frac{\pi \Delta p 16 m^4}{8 L \mu}$$

or

$$Q = \frac{2\pi m^4}{L \mu} \cdot \Delta p$$

For fluid flowing through a given section $\frac{2\pi m^4}{L}$ is constant.

Hence, let $\frac{2\pi m^4}{L} = a'$

Where a' is a constant, then $Q = \frac{a'}{\mu} \Delta p$

This proves that when the flow is entirely viscous or streamlined, then flow is directly proportional to the pressure drop. This can be written

as $\Delta p = a'' \mu Q$

where a'' is a constant.

CHAPTER V

THE DESIGN OF THE VISCOUS FLOW ELEMENT

The viscous flow element chosen for this design consists of a honeycomb of long narrow triangular passages 5.2" long, each having a cross sectional area of 0.0019 square inches.

The honeycomb element was formed by use of two long strips of aluminum, .005 in. thick and 5.2 in. wide, one plain and the other corrugated. One strip was placed on top of the other and the pair of sheets were then wound around a central core. It thus formed a cylindrical element having triangular passages with a length of 5.2 in. parallel to the axis of the cylinder.

The corrugated aluminum strip is stock material, used in a certain model of oil filter manufactured by the Fram Corporation. The availability of this material from the Fram Corporation was the reason it was selected for use in this design.

The size of the triangular sections of the passages formed by the aluminum sheets is shown below. The dimensions were obtained by taking a number of impressions of the triangular passageways on paper and averaging the measurements.

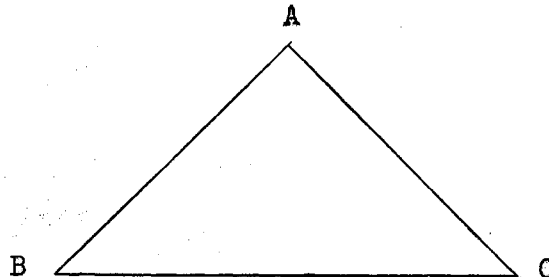


Figure 4. Section of the Triangular Passage

In Figure 4

$$AB = 1/16 \text{ in.} \quad AD = 0.0413 \text{ in.}$$

$$\text{perimeter} = 3/32 + 2/16 = 7/32 \text{ in.}$$

$$\text{area} = 0.0413 \times 1/2 \times 3/32 = .0019 \text{ sq. in.}$$

$$\begin{aligned} \text{hydraulic radius} &= \frac{\text{cross sectional area}}{\text{wetted perimeter}} \\ &= \frac{.0019}{7/32} = 0.0087 \text{ in.} \end{aligned}$$

Reynolds number may be used as an indicator of the probable velocity at which a change in the type of flow may be expected.

In engineering installations using actual pipe, it is usually found that the critical point corresponds to a Reynolds number of 1200. The following calculations based on a Reynolds number of 1200 will be used to establish the upper limit of the flow range in which laminar flow may be expected. Reynolds number may be calculated using the following equation.

$$R_e = \frac{DV\rho}{\mu}$$

where D is the diameter, equals 4 x hydraulic mean radius

μ is the viscosity, lbs sec per sq ft

V is the velocity, ft per sec

ρ is the density, slugs per cubic foot.

The viscosity of air at 80°F and 14.7 psia is

$$\mu = 3.87 \times 10^{-7} \text{ lbs sec per sq ft}$$

$$\rho = \frac{1}{v_g} = \frac{14.7 \times 144}{53.3 \times 540 \times 32.2}$$

$$= 2.28 \times 10^{-3} \text{ slugs per cubic foot}$$

$$D = 4 \times \frac{0.0087}{12} = 0.0029 \text{ ft}$$

$$R_e = \frac{DV\rho}{\mu} \quad \text{or} \quad V = \frac{R_e \mu}{D \rho}$$

$$V = \frac{1200 \times 3.87 \times 10^{-7}}{0.0029 \times 2.28 \times 10^{-3}} = 70.0 \text{ ft per sec.}$$

$$V = 70.0 \text{ ft per sec.}$$

It was tentatively assumed that the air meter capacity should be at least 7 cfs. This value is the estimated air flow requirement for a truck engine to be tested using the meter.

The area required for a Reynolds number of 1200, and velocity of 70.0 ft per sec is calculated as

$$A = \frac{Q}{V} = \frac{7}{70} \times 144 = 14.4 \text{ sq in}$$

The percentage of area taken up by the metal is 27.2%, hence

The total area is

$$1.27 \times 14.4 = 18.2 \text{ sq in}$$

and the diameter required for the viscous element, having a central core of 0.5 in. diameter, is calculated

$$\frac{\pi D^2}{4} - \frac{\pi d^2}{4} = \text{total area}$$

where D is the diameter of the viscous element

d is the diameter of the core

$$\frac{\pi}{4} (D^2 - d^2) = 18.3$$

$$D^2 - d^2 = \frac{18.3 \times 4}{\pi}$$

Thus D = 4.83 in.

In the case of air flow between parallel plates, it was found by Orr that turbulent conditions could exist at Reynold's numbers as low as 117.¹ Therefore, for the air meter capacity of 7 cfs it is desirable to base the viscous element diameter on a lower value of Reynold's number to insure laminar flow. Also the pressure drop across the viscous

1 Hubert O. Croft, Thermodynamics, Fluid Flow and Heat Transmission. McGraw Hill Book Company, Inc. New York, 1938.

element, is accurately measured at low Reynold's number.

Different values for diameters are determined corresponding to Reynold's numbers lower than 1200, and are tabulated in Table 1.

TABLE 1

Re No.	Velocity fps	Flow area sq in.	Total area sq in.	Diameter in	Differential Pressure	
					psf	in. of H ₂ O
100	5.85	173	220	16.7	3.77	0.725
120	7.04	144	207	16.2	4.55	0.875
140	8.20	123	156	14.05	5.30	1.02
160	9.35	108	137	13.2	6.05	1.16
180	10.5	96.2	122	12.4	6.80	1.31
200	11.7	86.2	109.2	11.7	7.56	1.45
400	23.4	43.2	55.0	8.38	15.20	2.92
600	35.2	28.7	34.2	6.60	22.8	4.38
800	46.8	21.6	27.2	5.88	30.3	5.82
1000	58.5	17.3	22.0	5.30	37.8	7.26
1200	70.0	14.4	18.3	4.83	45.2	8.70

The pressure drop across the viscous element is obtained by the formula

$$\Delta p = \frac{2\mu VL}{m^2}$$

where Δp is the pressure drop, lbs per sq ft

μ is the viscosity, lbs sec per sq ft

V is the velocity, ft per sec

L is the length of the viscous element, ft

m is the hydraulic mean radius, ft

For air at 80°F and 14.7 lbs per sq in.

$$\mu = 3.87 \times 10^{-7} \text{ lbs sec per sq ft}$$

$$V = 70 \text{ ft per sec at } Re = 1200$$

and for the viscous element

$$L = 5.2 \text{ in. or } 0.433 \text{ ft}$$

$$m = 0.00072 \text{ ft}$$

Thus substituting in the above formula

$$\Delta P = \frac{2 \times 3.87 \times 10^{-7} \times 0.433 \times 70}{(0.00072)^2}$$

$$= 45.2 \text{ psf}$$

$$= 8.70 \text{ in. of water.}$$

In order to use the meter for measuring larger air flow rates on engines other than the tested engines, it is desired to design the viscous element to have a larger diameter and corresponding smaller Reynold's number. However, the pressure drop should be large enough for satisfactory accuracy. With this subject in mind, and to make use of the material available, a diameter of 1 1/4 in. is selected for the viscous element.

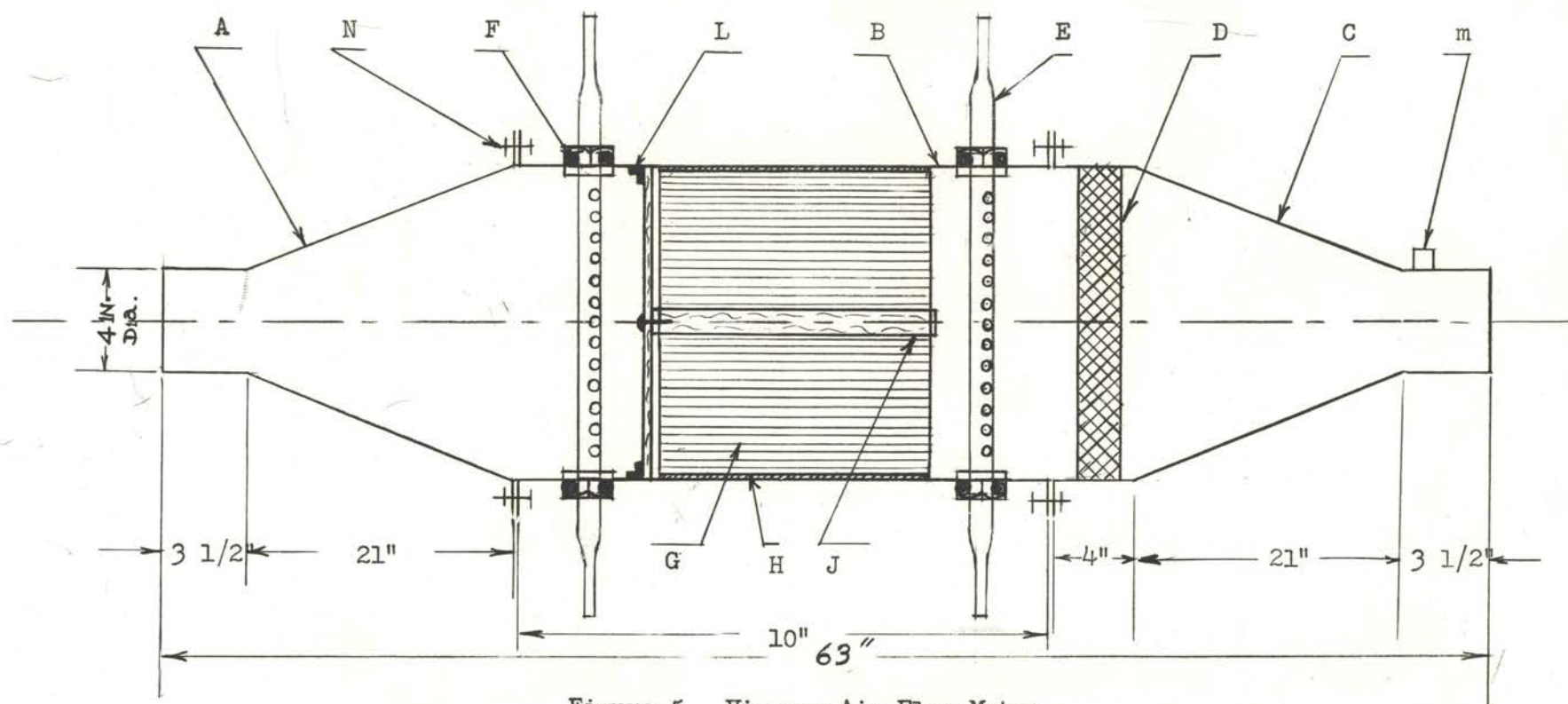


Figure 5. Viscous Air Flow Meter



Figure 6. Static Pressure Tube

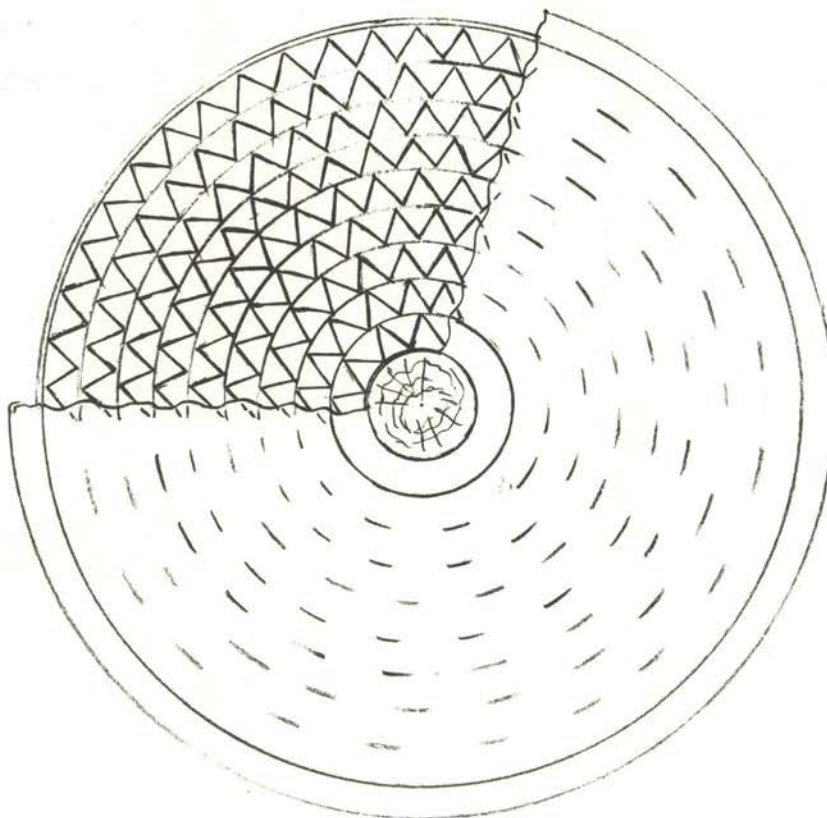


Figure 7. Sectional View of the Viscous Element

CHAPTER VI

DESCRIPTION OF THE APPARATUS AND EQUIPMENT

The housing of the viscous air-flow meter is made of galvanized iron. It is composed of three sections: the central cylindrical section (B) flanged at both ends, and the two end conical sections (A) and (C) having a flange at one end, tapering to a four inch diameter cylindrical section at the other end. The end section (C) is provided with short cylindrical section at the flanged end to house the filtering element.

The end sections (A) and (C) are fixed to the central section (B) by means of bolts. The central section (B) encloses the viscous element (G) and static pressure tubes (E).

A schematic view of the viscous air-flow meter is shown in Figure 5.

Filtering Element (D)

The filtering element (D) consists of a cylindrical element made of hardware cloth packed with copper turnings. It is soaked in a suitable filter oil, and the excess oil is drained off before it is assembled in position in the meter.

Viscous Element (G)

The viscous element (G) consists of a honeycomb of triangular passages, made by winding a pair of aluminum strips, one corrugated and the other plain, over a central wooden core. The diameter of the viscous element is 1 1/4 in. and the length is 5.2 in. This is assembled in section (B) in the space provided, and is fixed in position by means of a wooden strip (K) of 1/2 in. square section, set screw (I) and the small metallic angles (L) soldered to the housing. The

space between the element and the inside of the housing is packed with cotton to avoid the leaks. The assembly is shown in Figure 5.

Static Pressure Tubes (E)

The arrangement of the static pressure tubes is shown in Figure 5. These two metallic tubes are $1/2$ " inside diameter and 25 in. long, and fixed in the housing at right angles to each other. On these tubes small radial holes of $1/32$ in. in diameter are drilled in two rows making an angle of 60° with respect to each other, as shown in Figure 6. These holes are $1/2$ in. apart center to center. By this arrangement a true static positive pressure can be measured. In order to make air-tight connections between the tubes and the housing four sockets and packing nuts (F) carrying 'O' ring seals are provided.

Figure 6 shows the arrangement and the direction of the static pressure tubes.

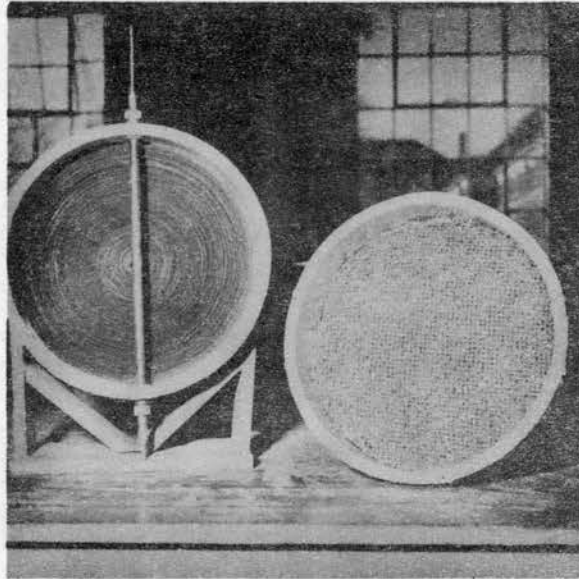
The whole assembly of the meter is fixed on a telescopic stand, by means of which the meter can be adjusted to any desired height. This whole arrangement can be seen in Figure 12 and 13.

A temporary collar and adapter arrangement is provided to install the different standard size nozzles which are used in the calibration of the meter. This arrangement is shown in Figure 13. At the inlet side of the meter a sleeve (m) is provided for inserting the thermometer in the air stream. Pressure tap joints and connections are made air tight with sealing compound. The pressure drop across the element was measured by means of a draft gage and standard water manometer. The manometer connections were made by means of rubber tubing. This arrangement is shown in Figures 10 and 13. The pressure taps at the

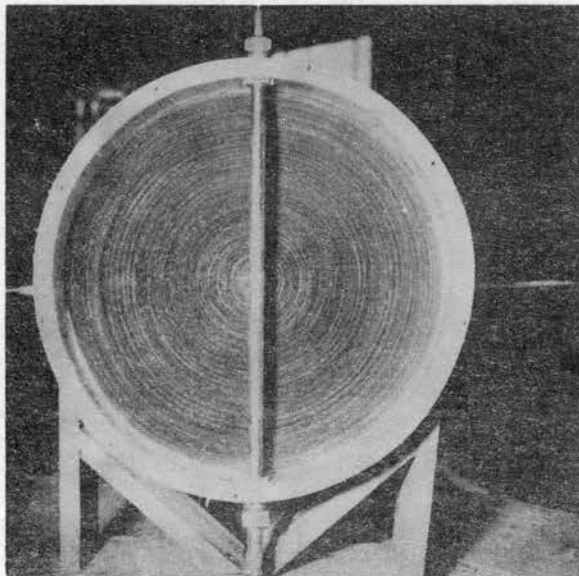
nozzle throats are made according to the specification given in ASME Power Test Code. A complete arrangement of the meter is shown in Figures 12 and 13.

Before the meter was calibrated, a check was made to see that all the connections and flange joints were air tight and the draft gage was level.

Also the mid point between the two rows of holes on both static pressure tubes should face upstream. This is to insure an accurate measurement of the pressure drop across the viscous flow element.



**Figure 8, VIEW OF THE FILTER AND VISCOUS
ELEMENT**



**Figure 9, CLOSE UP VIEW OF THE VISCOUS ELEMENT
SHOWING TRIANGULAR HONEYCOMB**

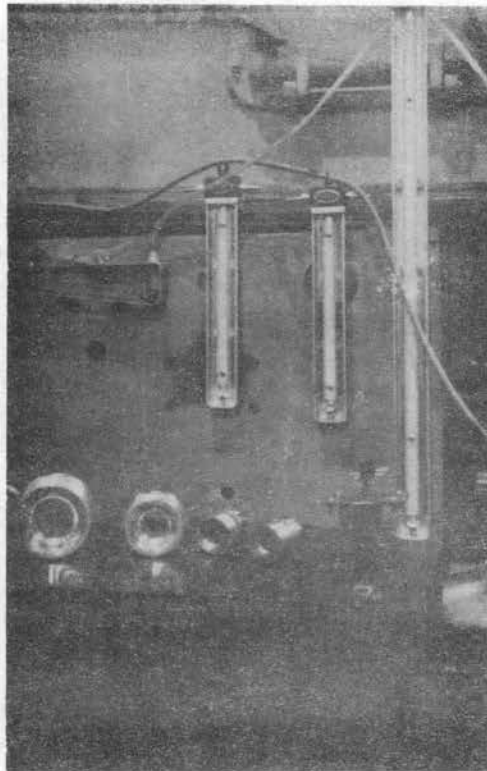


Figure 10, VIEW OF THE MANOMETERS

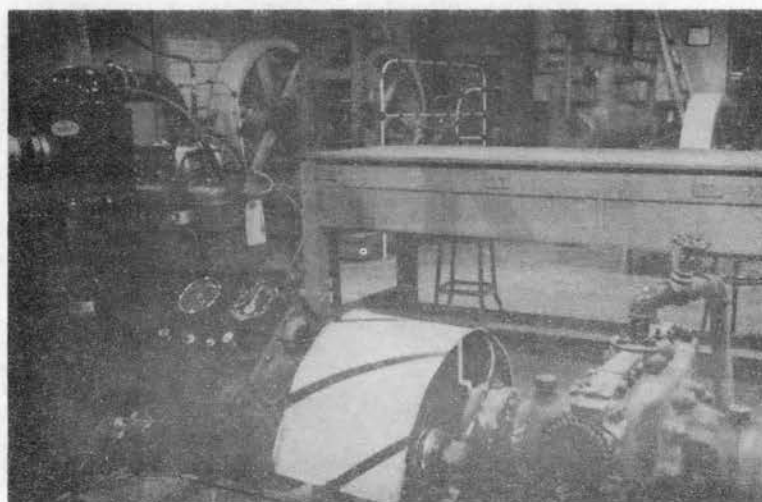
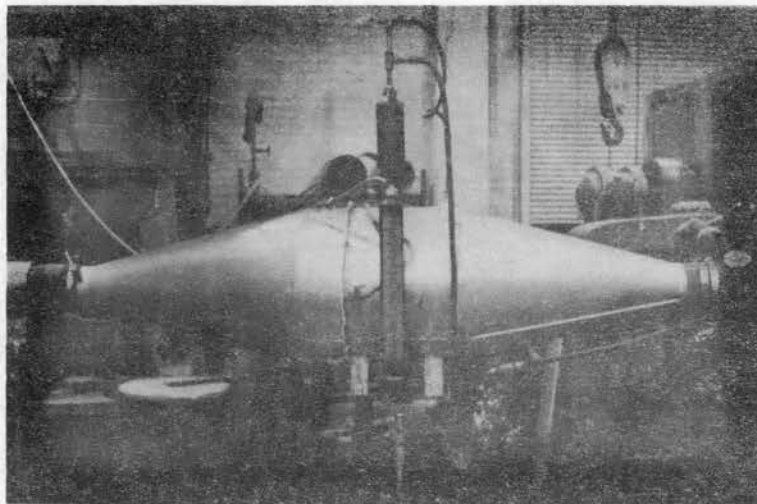
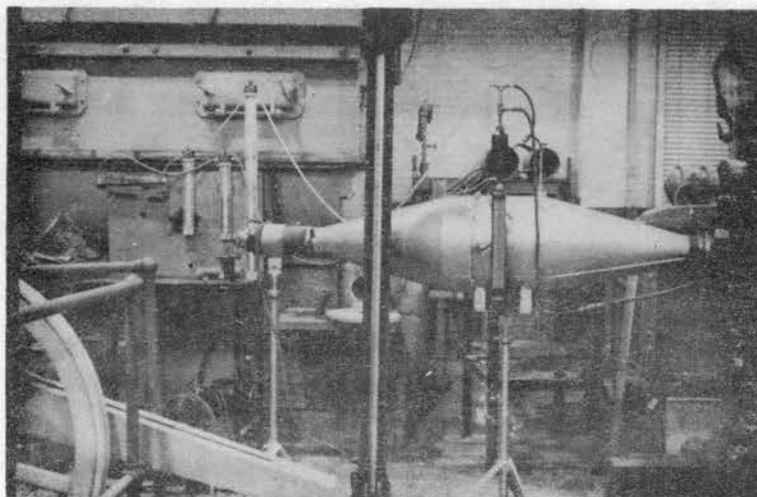


Figure 11, VIEW OF THE ENGINE TESTED



**Figure 12, ASSEMBLED VIEW OF THE VISCOUS
AIR FLOW METER**



**Figure 13, VIEW OF THE ARRANGEMENT OF APPARATUS AND
TESTING EQUIPMENT**

CHAPTER VII

PROCEDURE

The equipment for calibration was set up and connected to the engine as shown in Figure 13. The outlet of the air-flow meter was connected to the air cleaner inlet of the engine. The draft gage and manometer were leveled and the draft gage scale was adjusted to zero. The 13/8 in. nozzle was installed in the nozzle holder at the inlet end of the air meter and the manometer connections were made. After the preliminary check up of the engine to be used, the test was begun.

The engine was operated at various loads and speeds to obtain the desired rates of air flow through the nozzle and air meter. Simultaneous readings of the manometers were taken to obtain the pressure drop for the nozzle and the air meter at each of the selected test conditions. The operating conditions were controlled so as to vary the pressure drop for the nozzle within the range specified by the ASME Power Test Code. The same procedure was repeated for each of the following nozzle nozes: 1 in., 3/4 in. and 1/2 in.

The barometric pressure and the temperature of the air were observed several times during each group of tests and the average value of each was recorded for the test.

CHAPTER VIII

DATA

Observed Data

The data observed during the test are recorded in Tables 2 and 3 for different nozzles. These tables show the pressure drop across the nozzle p_N (in. of oil specific gravity .85) and corresponding pressure drop p_M across the meter (in. of water). The manometer readings were corrected to inches of water and recorded in the Tables of Calculated Data.

Calculated Data

The calculated data are recorded in Tables 4 and 5 for the different nozzles. The expansion factor (Y) used in calculating the flow through the nozzles is also included.

The results obtained from the observed and calculated data are shown in the following graphs:

Figure 14: Pressure drop across the nozzle versus pressure drop across the meter (from observed data).

Figure 15: Rate of flow versus the pressure drop across the meter (from the calibration data using 1 in, 3/4 in. and 1 3/8 in nozzle).

Figure 16: Rate of flow versus the pressure drop across the meter (from the calibration data using 1/2 in. nozzle).

Figure 17: Rate of flow versus the pressure drop across the meter at standard conditions (60°F and 14.7 psia).

Figure 18: Graph for the temperature correction.

Sample calculation is shown in the Appendix B.

OBSERVED DATA

Date: January 5, 1955

Time: 4:30 p. m.

Barometric Pressure: 29.16 in. of Hg.

Room Temperature: 76°F.

Specific gravity of manometer oil: .85

TABLE 2

Nozzle size in.	ΔP_N for nozzle in. of oil	ΔP_M for meter in. of water	Nozzle size in.	ΔP_N for nozzle in. of oil	ΔP_M for meter in. of water
3/4	11.8	0.055	1	2.30	0.04
	17.25	0.070		6.00	0.068
	20.5	0.072		8.00	0.082
	23.5	0.076		11.90	0.099
	28.0	0.081		13.90	0.105
	30.0	0.086		18.30	0.120
	32.0	0.091		21.30	0.130
	32.8	0.092		26.40	0.140
	40.7	0.100		27.80	0.150
	42.0	0.110		32.80	0.161
1 3/8			1 3/8	35.50	0.171
	1.0	0.06		7.6	0.145
	1.9	0.07		8.0	0.151
	3.5	0.096		9.4	0.160
	3.8	0.102		9.5	0.170
	3.9	0.105			
	6.1	0.123			
	6.2	0.130			

OBSERVED DATA

Date: January 6, 1955

Time: 4:30 p. m.

Barometric Pressure: 29.455 in. of hg.

Room Temperature: 77°F

Density of the Manometer Fluid: .85 for nozzle

TABLE 3

Nozzle size in.	Manometer Reading in. of H ₂ O		Manometer Reading in. of H ₂ O		
	ΔP_N for nozzle	ΔP_M for meter	Nozzle size in.	ΔP_N for nozzle	ΔP_M for meter
1/2	4.35	0.012	1/2	15.50	0.026
	6.80	0.016		17.00	0.027
	9.10	0.018		20.50	0.029
	10.7	0.020		22.50	0.031
	11.0	0.021		23.50	0.032
	13.2	0.023		26.60	0.035
	14.25	0.024		27.80	0.0351
	15.00	0.025		28.60	0.0352

CALCULATED DATA

Barometric pressure: 29.16" of Hg.

Room temperature: 76°F

Density of air: .0720 lbs/cu. ft at 70°F.

Discharge coefficient: .984 for the nozzle.

TABLE 4

Size of nozzle	Corrected ΔP_N in. of H_2O	ΔP_N lb per sq in.	Expansion factor Y	ΔP_M in. of H_2O	ΔP_M lb per sq in.	W lb per sec.
3/4	10.01	.36	.980	.055	.00198	.046
	14.66	.526	.977	.070	.00252	.056
	17.41	.626	.974	.072	.00259	.061
	20.00	.720	.972	.076	.00274	.065
	23.80	.855	.969	.081	.00292	.070
	25.50	.916	.962	.086	.00310	.072
	27.40	.982	.959	.091	.00328	.075
	27.90	1.005	.958	.092	.00331	.078
	34.60	1.245	.948	.100	.00360	.085
	35.70	1.285	.947	.110	.00396	.087
1	1.95	.070	1.000	.04	.00144	.037
	5.10	.184	.994	.068	.00259	.057
	6.8	.244	.982	.082	.00295	.068
	10.1	.364	.979	.099	.00256	.083
	11.8	.425	.982	.105	.00396	.090
	15.5	.560	.979	.120	.00432	.102
	18.1	.651	.973	.130	.00468	.110
	22.4	.806	.968	.140	.00505	.120

Size of nozzle in.	Corrected ΔP_N in of H_2O	ΔP_N lb per sq in.	Expansion factor Y	ΔP_M in of H_2O	ΔP_M lb per sq in.	W lb. per sec.
1	23.6	.850	.996	.15	.00540	.124
	27.8	1.000	.959	.161	.00580	.134
	30.2	1.090	.954	.171	.00615	.145
1 3/8	.85	.031	1.03	.06	.00216	.047
	1.615	.058	1.03	.07	.00252	.065
	2.97	.107	1.00	.096	.00346	.086
	3.23	.116	1.01	.102	.00367	.091
	3.32	.119	1.01	.105	.00378	.092
	5.26	.189	.997	.130	.00468	.114
	5.19	.187	.999	.123	.00442	.106
	6.45	.232	.990	.145	.00522	.126
	6.80	.244	.989	.151	.00543	.129
	8.00	.288	.984	.160	.00575	.138
	8.06	.290	.983	.170	.00612	.140

Barometric Pressure: 29.455 in. of Hg = 14.45 lbs per sq in.

Room Temperature: 77°F

Density of Air: .0726 lbs per cu ft.

Discharge Coefficient: .970

Density of the Manometer Fluid: .85 for nozzle

TABLE 5

Size of nozzle in.	ΔP_N lb per sq in.	Corrected ΔP_N in. of H ₂ O	Expansion Factor Y	ΔP_M in. of H ₂ O	ΔP_M lb per sq in.	W lb per sec.
1/2	.133	3.70	.991	.012	.000432	.0107
	.208	5.78	1.000	.016	.000575	.0135
	.278	7.74	.987	.018	.000648	.0155
	.328	9.10	.986	.020	.000720	.0167
	.336	9.35	.985	.021	.000756	.0172
	.403	11.20	.984	.023	.000828	.0186
	.435	12.10	.982	.024	.000865	.0192
	.437	12.75	.980	.025	.000900	.0193
	.475	13.20	.979	.026	.000935	.0200
	.520	14.42	.978	.027	.000972	.0210
	.626	17.40	.974	.029	.00104	.0230
	.688	19.10	.972	.031	.00111	.0240
	.716	19.90	.971	.032	.00115	.0244
	.810	22.60	.969	.035	.00126	.0260
	.850	23.60	.965	.0351	.001262	.0264
	.875	24.30	.964	.0352	.001270	.0268

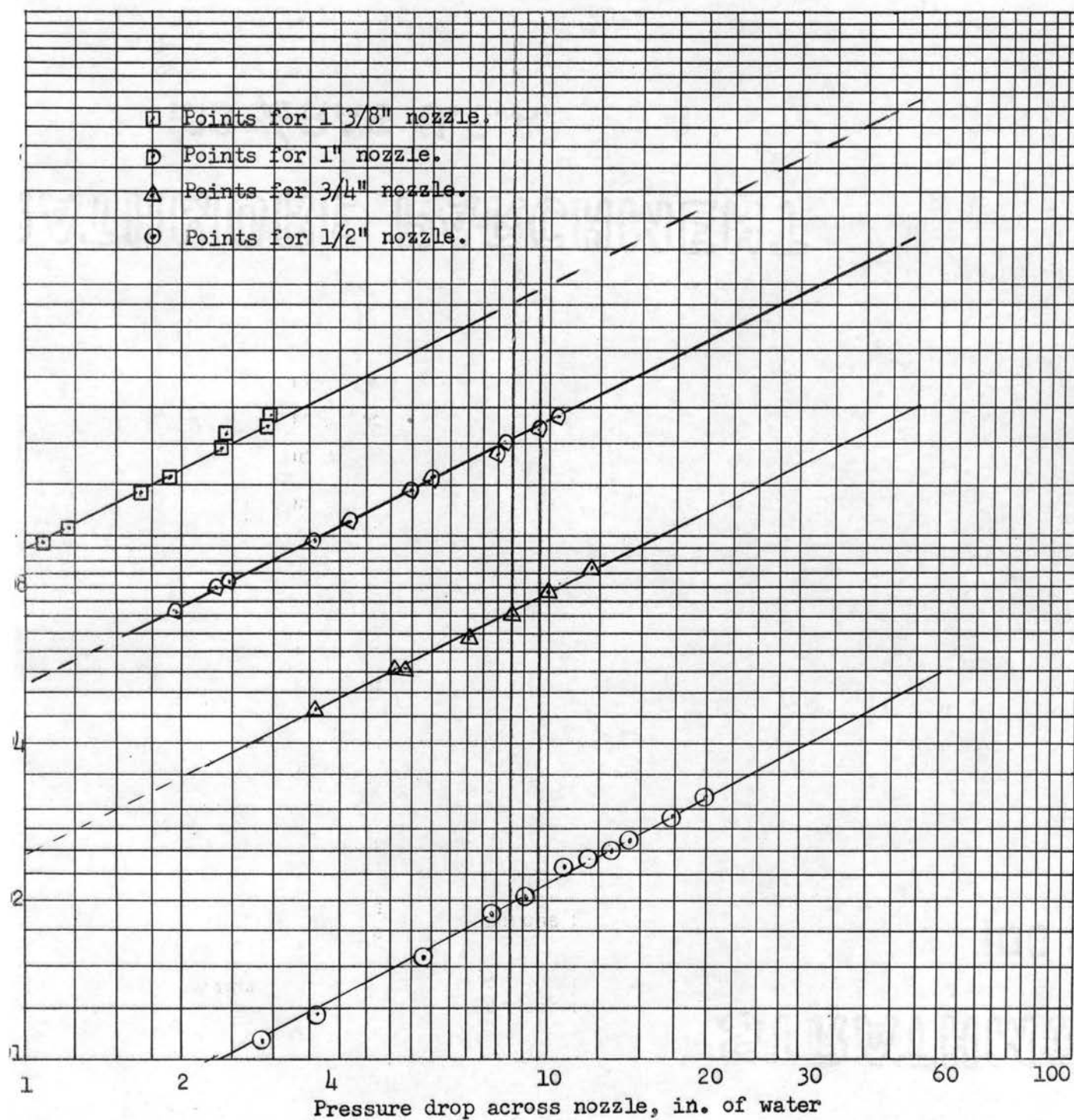


Figure 14. Pressure Drop Across Air Flow Meter Versus Pressure Drop Across Nozzles.

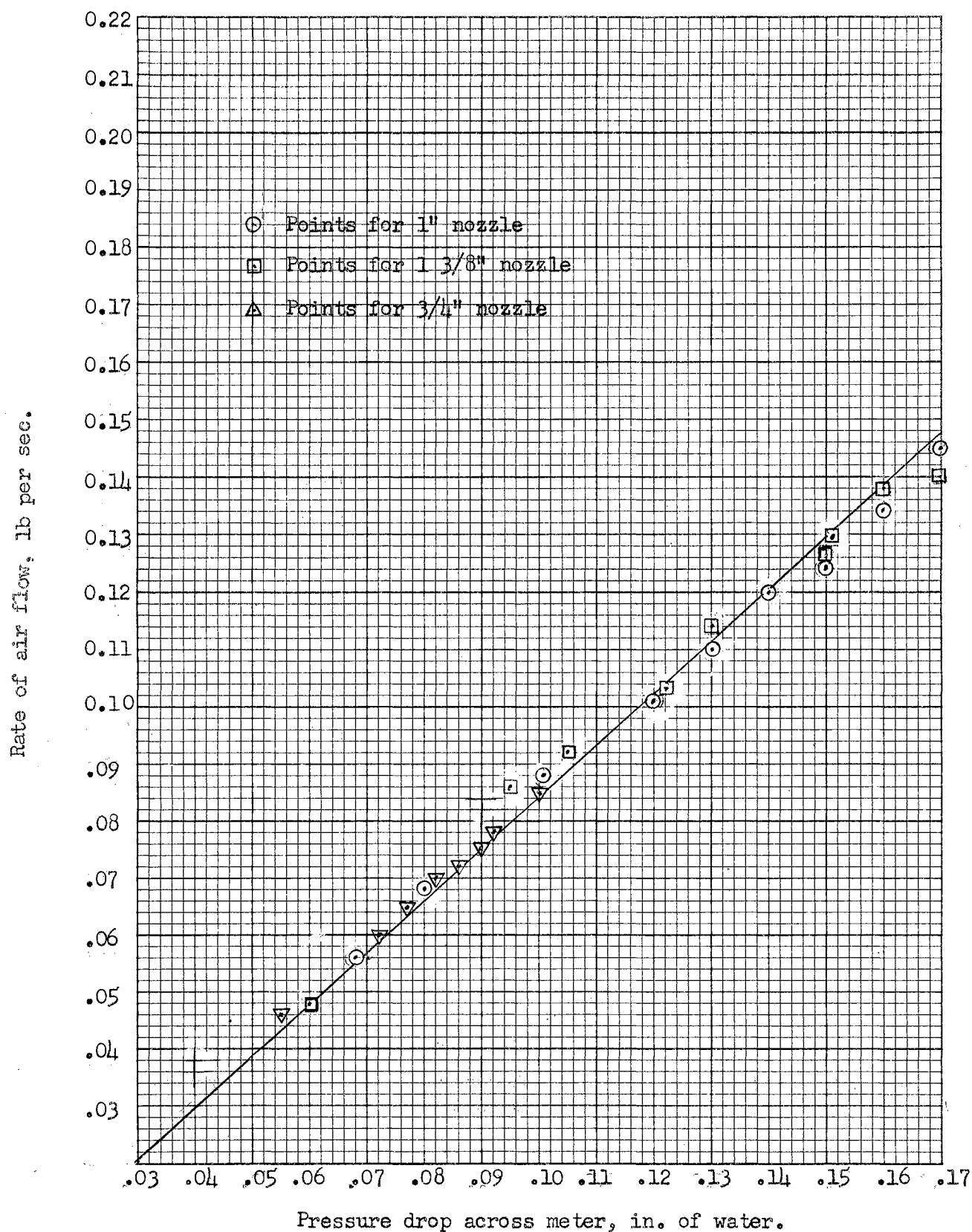


Figure 15. Calibration Graph for Meter using 3/4 in., 1 in., and 1 3/8 in. nozzles at 76°F and 14.30 psia.

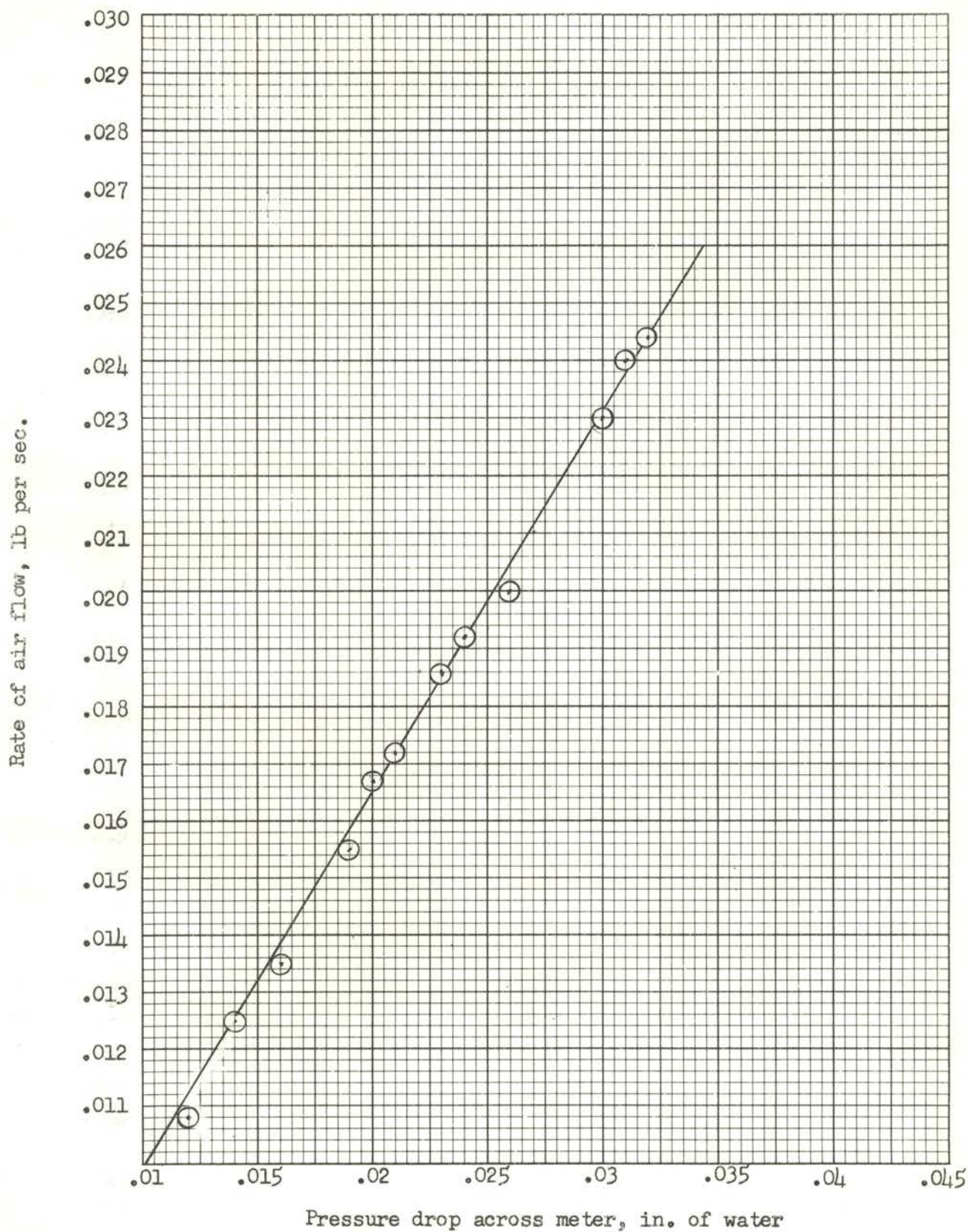


Figure 16. Meter Calibration Graph using 1/2 in. nozzle at 77°F. and 14.45 psia.

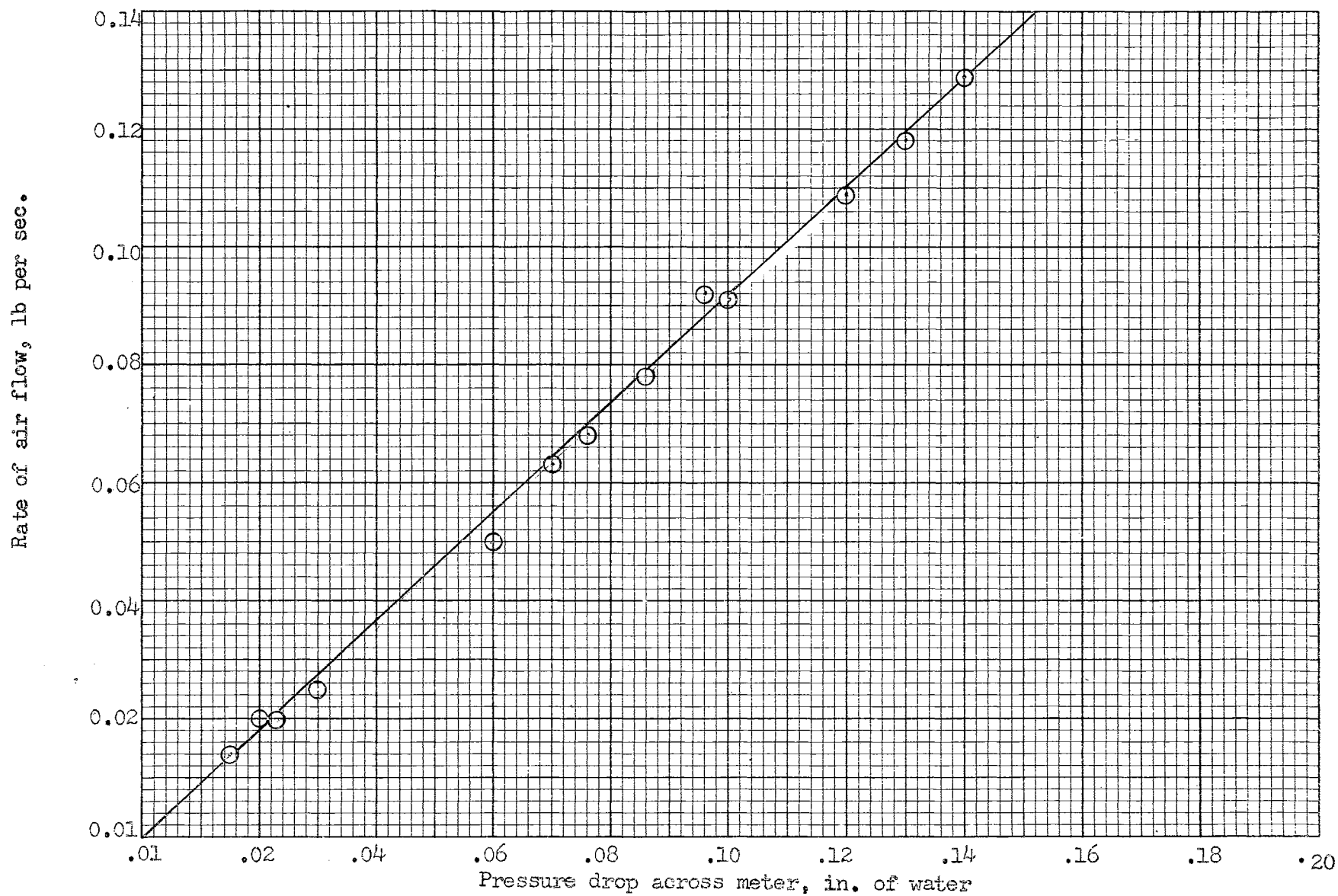


Figure 17. Meter Calibration Graph at Standard Conditions
[60° F. and 14.7 psia]

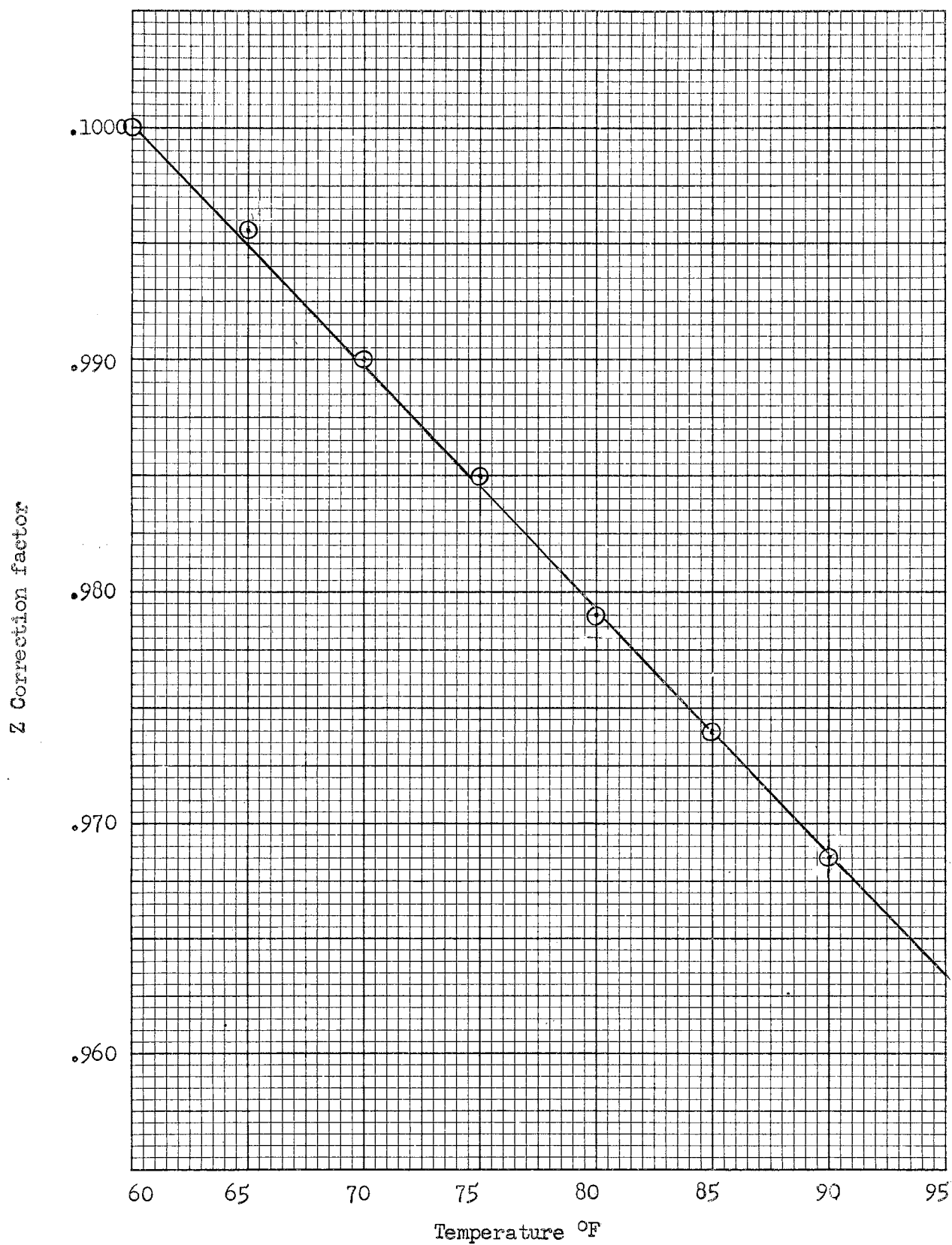


Figure 18. Temperature Correction Graph

CHAPTER IX

SUMMARY AND CONCLUSIONS

According to the theory on which the design of the viscous air flow meter was based, the pressure drop that occurs during laminar flow is directly proportional to the rate of flow of air.

The results of the tests as presented in Figure 15, 16, and 17 show a linear relationship between the pressure drop across the meter and the flow. This indicates that laminar flow was obtained for all rates of flow used during the tests. The meter was calibrated, by measuring the rate of flow through ASME standard nozzles of different sizes. Figure 14 shows that the pressure drop across the nozzle versus the pressure drop across the meter plotted on log paper is a straight line with a slope of approximately $1/2$. This also indicates that laminar flow through the meter was obtained since the theoretical relationship indicates that the graph should be a straight line with a slope of $1/2$. From the observed and calculated data, a calibration graph, Figure 17, for the meter at standard conditions was obtained.

The flow rate (W) at temperatures other than standard temperature may be obtained by use of the correction factor (Z) from Figure 18. The flow rate (W) must be multiplied by an additional correction factor $\frac{\text{barometer reading psi}}{14.7}$ to correct for the effect of atmospheric pressure. The size of the viscous element was made large enough to handle the total capacity of air flow to an engine considerably larger than the one used for the tests.

Conclusion

From the data obtained and calibration graphs, it was established that the flow of air through the meter was laminar. From the data obtained for the pressure drops on the nozzles it is seen that for the 1/2" nozzle the pressure drop measurements were not obtained very accurately. This is due to the fact that as the air flow increased the manometer readings fluctuated considerably. This indicates the presence of pressure pulsations which would affect the accuracy of flow measurement.

From the various graphs and calculated data it was established that the capacity of viscous element was underestimated. However, the small pressure drop resulting from the large capacity may be an advantage. For internal combustion engines testing it is desirable that the pressure drop across the meter be no larger than is necessary for satisfactory accuracy in the measurement of the pressure drop.

From the graphs it is seen that almost all of the experimental points are either on or close to the line through the average of the points. This indicates that satisfactory accuracy was obtained in taking the readings. The rates of flow obtained on the tests are considerably larger than the calculated value obtained from the theoretical equation on which the design was based. This equation contains a constant based on experimental results. It seems possible that the constant might have a different numerical value for the size and shape of flow passages used in this design. It also seems possible that the high velocity and turbulence associated with the air issuing from the nozzles affected the rate of flow.

Recommendations for Future Research

Additional tests and investigation might be undertaken to establish the maximum flow capacity of the meter within the laminar flow range.

This would require a larger engine or other machine to produce a larger flow rate. It would also be desirable to determine why the flow obtained from theory used in the design does not agree with the value obtained from the experimental results.

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APPENDIX A

FLOW OF FLUID THROUGH A NOZZLE

The equation presented in this section is for the flow of fluids through a well shaped ASME standard nozzle in which it is assumed that the outlet pressure will always be more than 53% of the inlet pressure.

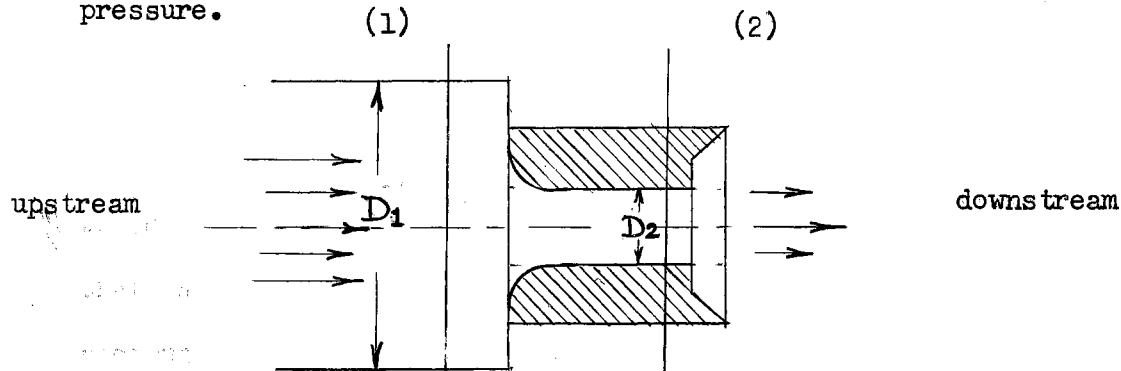


Figure 1. Section of the Nozzle

A section view of an ASME standard nozzle is shown in the above figure.

In deriving the formula for flow through such a nozzle, the following assumptions are made. The flow of air through the nozzle is assumed to be steady, and isentropic. The fluid flowing conforms to the law $pv = wRT$ and C_v remains constant. If these conditions hold true, the energy per lb of fluid will be the same at both sections 1 and 2, and the actual rate of flow is given

$$W = V_2 A_2 \rho_2 \text{ (referring to section 2)}$$

where W is the actual rate of flow, lb per sec

V_2 is the velocity of air ft per sec

ρ_2 is the density of air lb per cu ft.

$$\text{Hence } W = A_2 \frac{1}{\sqrt{1 - (D_2/D_1)^4}} \sqrt{2 g \rho \Delta p}$$

and W (actual) = CW (theoretical)

where C is the coefficient of discharge. The value of C is obtained from experimental results. This is sometimes combined with the velocity of approach factor

$$K = \frac{C}{\sqrt{1 - (D_2/D_1)^4}}$$

where K = flow coefficient.

The nozzle inlet is exposed to the atmosphere and hence K is equal to C as D_1 is infinite.

$$\text{Therefore } W = A_2 C \sqrt{2 g \rho_1 \Delta p}$$

where ρ_1 is the density of the fluid at section (1). This equation is used for liquids only, and for compressible fluids like a gas or air, isentropic flow being considered, a factor known as the expansion factor (Y) and also the factor (E) known as area multiplier for the thermal expansion of the nozzle to be taken into consideration.

Hence a general formula for the rate of flow for a compressible fluid flowing through a nozzle is given as

$$W = A_2 K E Y \sqrt{2 g \rho_1 \Delta p} \quad 1$$

where

$$Y = \left[\frac{k}{k-1} (P_2/P_1)^{\frac{2}{k}} \frac{1 - (P_2/P_1)^{\frac{k-1}{k}}}{1 - P_2/P_1} \right]^{1/2} \times \left[\frac{1 - (D_2/D_1)^4}{1 - (D_2/D_1)^4 (P_2/P_1)^{2/k}} \right]^{1/2}$$

k is the ratio of specific heats

P_1 and p_2 are the pressure

of air at sections (1) and (2) respectively.

In the above equation D_1 is infinite in the present application, hence the second factor of the equation is unity, and

1 "Flow Measurements", ASME Power Test Code, 1949.

$$Y = \left[\frac{k}{k-1} \left(\frac{p_2}{p_1} \right)^{\frac{2}{k}} \frac{1 - (p_2/p_1)^{\frac{k-1}{k}}}{1 - p_2/p_1} \right]^{\frac{1}{2}}$$

The velocity of approach factor k and thermal expansion factor E and are negligible; thus (k) becomes equal to the coefficient of discharge (C) and the equation for the rate of flow is

$$W = .668 A_2 C Y \sqrt{\rho_1 \Delta p}$$

This equation was used in the calculation of the flow rate through the nozzle

APPENDIX B

SAMPLE CALCULATIONS

Barometric pressure = 29.455 in. of Hg

$$= 29.455 \times .491 = 14.45 \text{ psi}$$

Room temperature = 77°F or 537°R abs.

Density of air at 77°F = 0.726 lb per cu ft.

Discharge coefficient = .970 for 1/2 in nozzle (ASME Code).

Pressure drop across the 1/2 in nozzle

$$\Delta P_N = 13.20 \text{ in of H}_2\text{O}$$

$$= 13.20 \times .036 = .475 \text{ lb per sq in.}$$

The gravimetric rate of flow of air through the nozzle is given as

$$W = .688 ACY \sqrt{\rho_1 \Delta P} \text{ lb per sec}$$

where A = area of the nozzle throat, sq in.

C = coefficient of discharge for the nozzle

Y = expansion factor

$$= \left[\frac{k}{k-1} (p_2/p_1)^{2/k} \frac{1 - (p_2/p_1)^{(k-1)/k}}{1 - p_2/p_1} \right]^{1/2}$$

k = 1.39 Ratio of specific heat of air

p₂ = pressure downstream, lb per sq in.

p₁ = pressure upstream, lb per sq in.

$$\Delta p = p_1 - p_2, \text{ lb per sq in.}$$

= density of the air, lb per cu ft. Nozzle throat area (A)

$$A = \frac{\pi D^2}{4} = \frac{\pi}{4} \times \frac{1}{4} = 0.196 \text{ sq in.}$$

where D is the nozzle throat diameter.

Expansion factor

$$Y = \left[\frac{k}{k-1} \left(\frac{P_1 - \Delta P}{P_1} \right)^{2/k} \frac{1 - \left(\frac{P_1 - \Delta P}{P_1} \right)^{\frac{k-1}{k}}}{1 - \left(\frac{P_1 - \Delta P}{P_1} \right)} \right]^{1/2}$$

substituting the values, in this equation

$$Y = \left[\frac{1.39}{.39} \left(\frac{14.45 - .475}{14.45} \right)^{2/1.39} \frac{1 - \left(\frac{14.45 - .475}{14.45} \right)^{\frac{1.39-1}{1.39}}}{1 - \left(\frac{14.45 - .475}{14.45} \right)} \right]^{1/2}$$

$$= \left[3.56 (.967)^{1.44} \frac{(1-0.967)^{.28}}{(1-0.967)} \right]^{1/2}$$

$$= \left[3.56 \times .952 \times \frac{0.0093}{0.033} \right]^{1/2} = \left[0.957 \right]^{1/2}$$

$$= 0.978$$

$$W = 0.668 \times .196 \times 0.970 \times 0.978 \sqrt{0.0726 \times 0.475}$$

$$= 0.0230 \text{ lb per sec.}$$

At this rate of flow the pressure drop across the viscous element was 0.026 in of water. Calculation for the temperature, pressure and viscosity correction is as follows:

$$\Delta P = \frac{2 \mu LV}{m^2}$$

at standard conditions of 60°F and 14.7 psia

$$\Delta P_{60} = \frac{2L}{m^2} \mu_{60} V_{60} \text{ and at any other temperature } t^{\circ}\text{F}$$

$$\Delta P_t = \frac{2L}{m^2} \mu_t V_t$$

assuming $\Delta P_{60} = \Delta P_t$

$$\mu_{60} V_{60} = \mu_t V_t \text{ or } V_{60} = \frac{\mu_t}{\mu_{60}} V_t$$

Therefore $W_{60} = Q_{60} \times \rho_{60} = A V_{60} \rho_{60}$ and

$$W_t = Q_t \rho_t = A V_t \rho_t$$

$$\text{and } \rho = \frac{P}{RT} \text{ or } \frac{\rho_{60}}{\rho_t} = \frac{P_{60}}{T_{60}} / \frac{P_t}{T_t}$$

$$\text{Thus } \frac{\rho_{60}}{\rho_t} = \frac{p_{60}}{p_t} \times \frac{T_t}{T_{60}}$$

$$\frac{W_{60}}{W_t} = \frac{\mu_t}{\mu_{60}} \times \frac{p_{60}}{p_t} \times \frac{T_t}{T_{60}}$$

$$\text{or } W_{60} = W_t \times \frac{\mu_t}{\mu_{60}} \times \frac{14.7}{p_t} \times \frac{T_t}{520}$$

For a temperature of 77° F and pressure of 14.45 lb per sq in.

$$W_t = W_{77} = 0.023 \text{ lb per sec}$$

$$\mu_t = \mu_{77} = .0185 \text{ centipoise and } \mu_{60} = .0180 \text{ centipoise}$$

$$W_{60} = 0.023 \times \frac{.0185}{.018} \times \frac{14.7}{14.45} \times \frac{537}{520}$$

$$= 0.023 \times 1.08$$

$$= .0248 \text{ lb per sec.}$$

$$W_t = \frac{\mu_{60}}{\mu_t} \times \frac{p_t}{14.7} \times \frac{520}{T_t} \times W_{60}$$

$$= Z W_{60}$$

Where Z is the correction factor.

$$Z = \frac{\mu_{60}}{\mu_t} \times \frac{p_t}{14.7} \times \frac{520}{T_t}$$

Hence correction factor Z is obtained for a temperature of 77°F and 14.7 lb per sq in.

$$Z = \frac{W_t}{W_{60}} = \frac{1}{1.08} = 0.926$$

$$\text{Hence } W_t = W_{60} \times 0.926.$$

If the pressure drop Δp_m across the meter is known, the corresponding W_{60} may be obtained from the graph. The correction factor at the desired temperature is obtained from the correction factor graph Figure 18 and hence the true rate of flow at Δp_m may be obtained corrected for pressure temperature and viscosity as shown above. Additional correction factor (Barometer reading psia divided by 14.7) to be multiplied by rate of flow for pressure correction.

VITA

S. Vicar Husain
candidate for the degree of
MASTER OF SCIENCE

Thesis: DESIGN AND CALIBRATION OF A VISCOUS AIR FLOW METER

Major: Mechanical Engineering (power)

Biographical and Other Items:

Born June 15, 1928 at Hyderabad Dn. India

Undergraduate Study: Bachelor of Science, Nizams College
Hyderabad, 1947. Bachelor of Science in Mechanical
Engineering, Osmania University, Hyderabad Dn. India,
1949.

Graduate Study: Oklahoma A & M College, 1953-1955.

Experience: Two years of industrial training, sponsored by
the Government of India, at Hindustan Aircraft Factory,
India and Cooper Engineering Ltd. (Diesel Engine Manu-
facture), Satara, India, (one year at each).
One year part time teaching at Oklahoma A & M College
for the Physics Department.

Member of the American Society of Mechanical Engineers, and Society
of Automotive Engineers.

Date of Final Examination: May, 1955

THESIS TITLE: DESIGN AND CALIBRATION OF A VISCOUS AIR FLOW METER

AUTHOR: S. Vicar Husain

THESIS ADVISER: Professor W. H. Easton

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TYPIST: Sara Preston